(54) Title: ECCENTRIC PLANETARY DRIVE TRANSMISSION
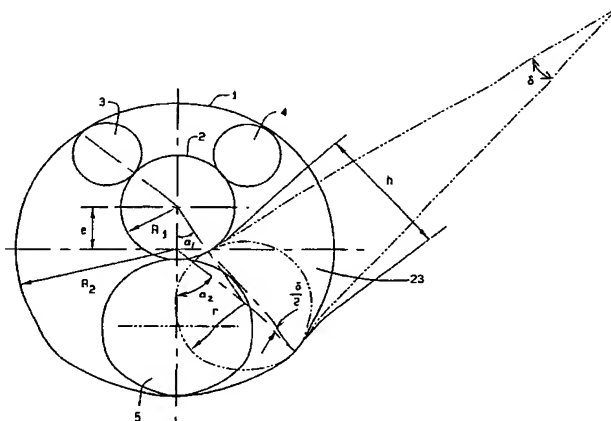
(57) **Abstract:** An eccentric planetary traction drive transmission (A) which includes at least two planetary rollers (3 & 4), sun roller member (2), and a carrier member (27). One of the planetary rollers (3, 4, or 5) is flexible and is positioned between and in contact with an outer ring member (1) and the sun roller member (2). Rotation of either the outer ring member (1) or the sun roller member (2) wedges the flexible planetary roller (3, 4, or 5) within a convergent wedge gap (23) which squeezes the flexible planetary roller (3, 4, or 5) between the outer ring member (1) and the sun roll member (2). Friction between the flexible planetary roller (3, 4, or 5), the sun roller member (2), and the outer ring member (1) transmits rotational motion and torque between the outer ring member (1) and the sun roller member (2). The other at least one supporting planetary roller (3, 4, or 5) is a supporting roller which supports the sun roller member (2) and the carrier member (27). A plurality of bearings (26) supports the sun roller member (2) within the outer ring member (1) and the at least one supporting planetary roller (3, 4, or 5).

ECCENTRIC PLANETARY DRIVE TRANSMISSION

MECHANISM

<u>Technical Field</u>

5   This invention relates in general to a planetary traction drive transmission, and, more particularly, to a planetary traction drive transmission with at least one flexible roller having an adaptive self-loading mechanism.

<u>Background Art</u>

        Traction drives use frictional force to transmit torque and power.

10   Because the power is transmitted between two smooth surfaces, often through a thin layer of lubricant, a traction drive possesses unique characteristics that are not readily attainable by gear drives. These characteristics include quietness, high-efficiency, high rotational accuracy, and zero-backlash.

        Generating adequate normal force at the contact is essential for traction

15   drives. Various loading mechanisms have been proposed. These mechanisms have lead to a host of designs. A common practice is to use tapered surfaces along the axial direction. By moving these surfaces axially, a radial displacement and thus normal force are generated. Examples of such designs are disclosed in U.S Patent Numbers 3,475,993 and 3,375,739.

20   Since the envelopes of the tapered surfaces in most designs do not necessarily converge to a common point, this results in a so-called spin motion at contacting surfaces. The spin motion not only offsets the high-efficiency otherwise provided by the traction drive, but also causes component wear and high break away torque.

25   Recently, a design of zero-spin planetary traction drive has been proposed by Ai as disclosed in the U.S. Patent Number 6,095,940. This design employs the on-apex concept similar to that of tapered roller bearings. Two rows of planetary rollers are used to balance the internal axial force on the planetary rollers. Although this design offers a torque actuated loading

30   mechanism and greater torque capability, it is somewhat complex in construction.

The cylindrical planetary traction drive is also able to achieve zero-spin motion. However, generating sufficient normal force at the contacts has been a challenge. Designs proposed in the past have offered various means to pre-load the drive either by mechanically deforming the outer rings or by thermal assembling the drive. The pre-load generated by such means, in general, can not be adjusted during operation. For partial load application, traction drives are unnecessarily overloaded. This has a negative impact on transmission efficiency and service life.

Perhaps the simplest means to generate torque responsive load is using eccentric planetary drives as was disclosed by Dieterich U.S. Patent Number 1,093,922 in 1914. Over the years, various improvements have been proposed. See for example, U.S. Patent Numbers 3,945,270, 4,481,842, 4,555,963, and foreign patent numbers JP10-311398, EP 0,856,462 A2. They all have multiple planetary rigid rollers, and each planetary roller requires a supporting shaft.

While friction actuated loading mechanisms have been disclosed in prior art where one of the planetary rollers was entrained by friction force into the wedged-space created by eccentric raceways, the loading roller was rigid and thus the entrainment angle was kept constant. This was based on the notion that the maximum traction coefficient was constant during the operation. The entrainment angle was determined based on a specified friction coefficient, which was assumed to be not a function of the contact load. Such a loading mechanism was either over conservative or inadequate when the maximum available traction coefficient changes with contact load.

For eccentric planetary drives, the planetary rollers have different sizes. In most of the prior art designs, this has led to different contact stresses at the contact between the sun roller and planets.

In some designs proposed in prior art, the drive had more than one loading roller that is moveable. This could cause a change in eccentricity between input and output shafts as the loading rollers adjusted their positions.

Therefore, it is desirable to provide a simple and improved design that allows for an adaptive, torque responsive loading mechanism where the entrainment angle changes as the load varies to match the variation in the

maximum available traction coefficient, a design that offers balanced contact stresses for all planet rollers, and a design that reduces or eliminates the change in eccentricity as the loading roller is adjusting its position.

Summary of the Invention

5   This patent application relates to planetary traction drives in general and specifically to an eccentric, cylindrical planetary traction drive having at least one flexible planetary roller capable of providing an adaptive self-loading mechanism to facilitate the transfer of rotational motion and torque. The current invention also provides a traction drive that allows for an adaptive

10   torque responsive loading and also provides a traction drive with well-balanced contact stresses at the most critical contacts.

Brief Description of Drawings

          FIG. 1 is a front view of the present invention.

          FIG. 2 is a left side sectional view of the present invention.

15   FIG. 3 is an oblique sectional view of the present invention.

          FIG. 4 is an exploded view of the present invention.

          FIG. 5 is a graphical representation of the mathematical relationship between various components of the present invention.

          FIG. 6 is a graphical representation of the normal forces between the

20   components of the present invention.

          FIG. 7 is a graph showing the relationship between the contact load and the geometry/traction coefficient of the present invention.

          FIG. 8 is a front view of a second embodiment of the present invention.

          FIG. 9 is a left side sectional view of a second embodiment of the

25   present invention.

          FIG. 10 is an oblique sectional view of a second embodiment of the present invention.

          FIG. 11 is an exploded view of a second embodiment of the present invention.

30   Corresponding reference characters indicate corresponding parts throughout the several views of the drawings.

Best Mode for Carrying Out the Invention

Referring now to Figures 1 and 2, one embodiment of the cylindrical planetary traction drive A comprises an outer ring member 1, a sun roller member 2, a first planetary roller 5, two second planetary rollers 3 & 4, and a carrier member 27. The outer ring member 1 further comprises a first cylindrical raceway 9 surrounding the axis of rotation, and a first fixed flange 10 and a second fixed flange 11.

The sun member 2 includes a second cylindrical raceway 12, a third fixed flange 13, a fourth fixed flange 14, and a shaft 15. The first planetary roller 5 has a third cylindrical raceway 16, and the second planetary rollers 3 and 4 have a fourth cylindrical raceway 18 and 17 (Figure 4) respectively. The first planetary roller 5 and the two second planetary rollers 3 and 4 are placed between and in contact with first cylindrical raceway 9 and second cylindrical raceway 12.

At least one of either the planetary roller 5 or the planetary rollers 3 and 4 is flexible compared to other rollers. In the embodiment shown herein, the first planetary roller 5 is the flexible roller and serves as a loading roller. Thus, it will hereafter be called the loading planetary roller 5. The loading planetary roller 5 deforms noticeably when diametrically squeezed.

At least one of the second planetary rollers 3 and 4 is a supporting roller. The supporting roller is relatively rigid and hardly deforms under diametrical load. In the embodiment shown herein, both planetary rollers 3 and 4 are supporting rollers and will hereafter be called the supporting planetary rollers 3 and 4. The supporting planetary rollers 3 and 4 are firmly fixed by shafts 22 and 25 respectively to the carrier member 27. Each of the supporting planetary rollers 3 and 4 is mounted onto its respective shaft 22 or 25 with a bearing 26 and two snap rings 24.

The carrier member 27 contains a base plate 6 and a cover plate 7. There are extrusions 19, 20, and 21 protruding perpendicularly from the back face of the base plate 6. The cover plate 7 is fastened to the top of the extrusions using fasteners 8, forming cavities for receiving the loading planetary roller 5 and the supporting planetary rollers 3 and 4. The first cylindrical

raceway 9 on the outer ring member 1 is eccentric to the second cylindrical raceway 12 on the sun member 2. The space between the first cylindrical raceway 9 and the second cylindrical raceway 12 forms a wedge gap 23 (Fig. 5).

The width of the wedged-gap h is expressed as:

5

$$h = 2r \cos \frac{\delta}{2} \qquad (1a)$$

It can be approximated by the following cosine function:

$$h \approx R_2 - R_1 + e \cos \alpha_1 \qquad (1b)$$

where R_1 is the radius of the sun roller raceway 12; R_2 *is* the radius of the first cylindrical raceway 9; and e represents the eccentricity between the raceways

10   12 & 9 of the sun roller member and the outer ring.

The wedge angle δ for planetary roller at azimuth position α_1 is given by:

$$\delta = Arc \cos \left[\frac{(R_1 + r)^2 + (R_2 - r)^2 - e^2}{2(R_1 + r)(R_2 - r)} \right] \qquad (2)$$

where r is the effective radius of the planetary roller at this azimuth position,

15   and can be expressed in terms of α_1 as:

$$r = \frac{R_2^2 - R_1^2 - e^2 + 2eR_1 \cos \alpha_1}{2(R_2 + R_1 - e \cos \alpha_1)} \qquad (3a)$$

The wedge angle δ represents a contact geometry condition.

The geometry coefficient μ_G is defined as:

$$\mu_G = \tan\left(\frac{\delta}{2}\right) \qquad (4)$$

20   As can be seen from equations (1) to (4), for α_1 between 0 and 90 (or 0 and -90), when α_1 moves away from the $\alpha_1 = 0$ position, the width of the gap h reduces, the wedge angle increases and thus geometry coefficient μ_G increases.

In the embodiment shown in FIG. 1, the flexible loading planetary roller 5 is assembled in the wedge gap 23 at an azimuth angle between $\alpha_1 = -90$ to 90

25   degrees, preferably in vicinity of $\alpha_1 = 0$. The center of the flexible loading planetary roller 5 is left floating along the wedge gap 23. The flexible loading planetary roller 5 is sufficiently flexible in the radial direction. When squeezed,

its effective diameter reduces in the corresponding direction. It is recommended that at home position, $\alpha_1 = 0$, the flexible loading planetary roller 5 is slightly squeezed.

5   The supporting planetary rollers 3 and 4 are arranged in wedge gap 23 generally between azimuth positions of $\alpha_1 = 90$ to 270 degrees. The centers of each of the supporting planetary rollers 3 and 4 are fixed to the carrier member 27.

The planetary traction drive A is preferred to operate with the carrier member 27 stationary. During operation, the traction force tangent to the third

10   cylindrical raceway 16 of the loading planetary roller 5 always drags the loading planetary roller 5 into the convergent wedge gap 23. The loading planetary roller 5 is thus squeezed, generating substantial contacting force normal to the contact surfaces. If the eccentricity e in relationship to geometry of the planetary train is favorable, a balance is achieved where the maximum available

15   traction force is equal to or greater than the operating traction force. This condition is called frictional self-loading. The relationship for ensuring such fictional self-loading is set forth by:

$$\mu_G \leq \mu_T \qquad\qquad (5a)$$

where μ_T is the maximum available friction coefficient at the contacts.

20   For optimal efficiency and service life of the planetary traction drive A, it is always desirable to have geometry coefficient μ_G close to but slightly smaller than the maximum available traction coefficient μ_T under various load conditions. That is:

$$\mu_G \lessapprox \mu_T \qquad\qquad (5b)$$

25   During operation, the loading planetary roller 5 is entrained into the convergent wedge gap 23 and squeezed. Consequently, the effective diameter (or radius) of the loading planetary roller 5 reduces. The loading planetary roller 5 thus moves to a new azimuth position, establishing a new balance. At the new position, the width of wedge gap 23 is narrower and the contact

30   geometry coefficient μ_G is increased.

The diametrical reduction of the loading planetary roller 5 can be estimated by:

$$2dr = \left(\frac{\pi}{4} - \frac{2}{\pi}\right)\frac{Wr^3}{EI} \tag{6}$$

where W is the contact load; E is the Young's elastic modulus; and I is the area moment of inertia of ring cross section for the flexible first planetary roller 5.

The geometry coefficient μ_G corresponding to this diametrical reduction is:

$$\mu_G = \tan\left\{\frac{1}{2} Arc\cos\left[\frac{(R_1 + R_2 + e - 2dr)^2 + (R_1 + R_2 - e + 2dr)^2 - 4e^2}{2(R_1 + R_2 + e - 2dr)(R_1 + R_2 - e + 2dr)}\right]\right\} \tag{7a}$$

With equations (6) and (7a), the increase in geometry coefficient μ_G with the contact load can be quantified. Figure 7 shows the variation of μ_G, with contact load W for planetary rollers with different cross section moments of inertia I.

The maximum geometry coefficient is bounded by

$$\mu_{G\max} = \tan\left\{\frac{1}{2} Arc\cos\left[1 - 2\left(\frac{e}{R_1 + R_2}\right)^2\right]\right\} \tag{7b}$$

On the other hand, a large body of research results (Tevaarwerk, NASA CR-165226, 1981) showed that the maximum available traction coefficient μ_T increases with contact pressure or contact load. The change in μ_T with contact load for a traction fluid is also plotted in Figure 7 with symbols and a dotted line.

The flexible, floating planetary roller design of the current invention thus provides an opportunity for an adaptive frictional self-loading mechanism. That is, to adaptively change contact geometry coefficient to match the change in maximum available traction coefficient. As a result, not only constrain (5a) can be met, but also the geometry coefficient can be kept close to the maximum available traction coefficient (constraint (5b)) for the entire load spectrum. Adaptive self-loading can be achieved by choosing a flexible planetary roller with adequate cross section moment of inertia I. With this flexible roller it is possible to the change contact geometry coefficient and have the change match

or compensate for the change in the maximum available traction coefficient when contact load changes. For example, the second curve in Figure 7 with $I =$ 13 mm^4 matches the change in maximum traction coefficient. This curve demonstrates the adaptive design principle.

The flexible loading planetary roller 5 can be as simple as a ring, however, other design alternatives are also possible.

The torque capacity of a traction drive is determined by the maximum allowable stress at the contact. Since planetary rollers for an eccentric planetary traction drive may have different sizes, the maximum contact stress for each planetary roller is not necessarily the same. To improve fatigue life, it is desirable to select the planetary roller size and place these rollers in proper azimuth positions so that every planetary roller has approximately the same contact stress.

Assume under a specified load (usually the maximum weighted load), the effective radius of the flexible loading planetary roller 5 is r_5 (after being squeezed), and its azimuth position is α_5 (see Figure 6). The radii of the supporting planetary rollers 3 and 4 are determined from the following equations in conjunction with equation (3).

$$\overline{R}_3 \sin(\alpha_5 - \alpha_3) + \overline{R}_4 \sin(\alpha_5 - \alpha_4) = 0 \qquad (8)$$

$$\overline{R}_3 \cos(\alpha_5 - \alpha_3) + \overline{R}_4 \cos(\alpha_5 - \alpha_4) + \overline{R}_5 = 0 \qquad (9)$$

where $\overline{R}_i = \dfrac{R_1 \cdot r_i}{R_1 + r_i}$ (i = 3, 4, or 5) represents the composite contact radii for contact between the sun roller 2 and the planetary rollers 3, 4, and 5, respectively.

The geometry relationship between the radius and azimuth position of any supporting planetary roller is given by equation (3). That is:

$$r_i = \frac{R_2^2 - R_1^2 - e^2 + 2eR_1 \cos\alpha_i}{2(R_2 + R_1 - e\cos\alpha_i)} \qquad (3b)$$

Subscript i = 3 and 4 refers to the supporting planetary rollers 3 and 4 respectively. To avoid edge stress, first cylindrical raceway 9 and second cylindrical raceway 12 are may be crowned. Or, alternatively, the third and

fourth cylindrical raceways 16, 17, and 18 of the planetary rollers 3, 4, and 5 may be crowned.

The outer surface of the outer ring member 1 can have gear teeth to communicate motion of rotation to a driving or driven component. Alternatively, the outer surface of the outer ring member can be made into a pulley to communicate motion of rotation to a belt.

A second embodiment of the current invention is shown in Figures 8 through 11. The second embodiment comprises an outer ring member 101, a sun roller member 102, two supporting planetary rollers 103 and 104, a flexible loading planetary roller 105, a carrier member 127, and a housing 130. The outer ring member 101 includes a first cylindrical raceway 109, and a shaft 133 supported on the housing 130 through a double row ball bearing 131. The sun roller member 102 has a second cylindrical raceway 112 and a shaft 115. The shaft 115 is supported on the carrier 127 by a ball bearing 132.

The second cylindrical raceway 112 on the sun roller member 102 is surrounded by and set eccentric to the first cylindrical raceway 109 in the outer ring member 101. The two supporting planetary rollers 103 and 104 are arranged in the wedge gap between the second cylindrical raceway 112 and the first cylindrical raceway 109. Each supporting planetary roller 103 and 104 has a shaft 122 that is fixed to the carrier member 127. The supporting rollers are supported on the shaft 122 by bearings 126. The flexible loading roller 105 is arranged in the wedge gap between the two supporting rollers. The center of the flexible roller is free to rotate along the wedge gap.

The carrier member 127 includes a base plate 106 and a cover plate 107. The base plate 106 has three protrusions 119, 120, and 121. The cover plate 107 is bolted to the protrusions on the base plate 106. The carrier member 127 is mounted to the housing 130 by a fastening means such as bolts. Bearing 131 is axially located by a snap ring 134 and a shoulder 136 in the housing 130. Bearing 132 is axially fixed by a snap ring 135 and a shoulder 137 in the carrier member 127.

Claims

1.   A planetary traction drive transmission comprising:

an outer ring member;

a carrier member;

a sun roller member eccentric to the outer ring member;

at least one loading planetary roller; and

at least one supporting planetary roller wherein one of either the at least

one loading planetary roller or the at least one supporting

planetary roller is sufficiently flexible in the radial direction to

communicate the rotational motion between the outer ring

member and the sun roller member and to effectively change the

geometry coefficient as load changes.

2.   The planetary traction drive transmission of claim 1 wherein the

outer ring member further comprises an axis of rotation and a first cylindrical

raceway surrounding the axis of rotation.

3.   The planetary traction drive transmission of claim 2 wherein the

sun roller member further comprises a first shaft and a second cylindrical

raceway, the second cylindrical raceway being eccentric to the first cylindrical

raceway to form a convergent wedge between the said first and second

cylindrical raceway.

4.   The planetary traction drive transmission of claim 3 wherein the

at least one loading planetary roller includes a third cylindrical raceway and

each of the at least one supporting planetary roller has a fourth cylindrical

raceway.

5.   The planetary traction drive transmission of claim 4 further

comprising at least one support shaft for supporting the at least one supporting

planetary roller in the carrier member.

6.   The planetary traction drive transmission of claim 5 wherein

the carrier member includes a base plate, a cover plate, and at least two

fasteners,

the base plate having at least two extrusions protruding from the back

surface of the base plate,

the at least two extrusions being sized and located on the base plate such
as to form cavities for receiving the at least one loading planetary
roller and the at least one supporting planetary roller, each of the
at least two extrusions also having a threaded hole therein,

the cover plate having a set of mounting holes located in the cover plate
such that each of the set of mounting holes aligns with one of the
threaded holes in the at least two extrusions, and

wherein the base plate has a first set of support shaft openings and the
cover plate has a second set of support shaft openings, both sets
of support shaft openings being sized and located to accept the at
least one support shaft for the at least one supporting planetary
roller.

7.   The planetary traction drive transmission of claim 6 wherein
each of the at least one supporting planetary roller is supported on its respective
support shaft by a bearing, and further comprising a means for retaining the
bearing onto the individual support shaft of each of the at least one supporting
planetary roller.

8.   The planetary traction drive transmission of claim 7 wherein the
at least one loading planetary roller is assembled in the convergent wedge gap at
an azimuth angle between α_l = -90 to 90 degrees and the center of the at least
one loading planetary roller is free floating along the convergent wedge gap.

9.   The planetary traction drive transmission of claim 8 wherein the
at least one supporting planetary roller is arranged in the convergent wedge gap
generally at an azimuth angle between α_l= 90 to 270 degrees and the center of
each of the at least one supporting planetary roller is fixed in relation to the
carrier member.

10.   The planetary traction drive transmission of claim 9 wherein the
wedge angle δ for the at least one loading planetary roller at azimuth position α_l
is characterized by

$$\delta = Arc\cos\left[\frac{(R_1+r)^2 + (R_2-r)^2 - e^2}{2(R_1+r)(R_2-r)}\right]$$

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where:

R_1 is the radius of the second cylindrical raceway of the sun roller member;

R_2 is the radius of the first cylindrical raceway of the outer ring member;

e represents the eccentricity between the first cylindrical raceway and the second cylindrical raceway; and

r is the effective radius of the at least one loading planetary roller at this azimuth position and is expressed in terms of α_l as:

$$r = \frac{R_2^2 - R_1^2 - e^2 + 2eR_1 \cos\alpha_1}{2(R_2 + R_1 - e\cos\alpha_1)}$$

11.   The planetary traction drive transmission of claim 10 wherein a geometry coefficient μ_G is set to vary with the contact load through change in the wedge angle δ to provide an adaptive loading mechanism and the geometry coefficient μ_G is characterized by

$$\mu_G = \tan\left(\frac{\delta}{2}\right)$$

12.   The planetary traction drive transmission of claim 11 wherein the variations in the geometry coefficient μ_G is induced mainly by a diametrical reduction of the at least one loading planetary roller as the at least one loading planetary roller is squeezed within the convergent wedge gap where the diametrical reduction is characterized by

$$2dr = \left(\frac{\pi}{4} - \frac{2}{\pi}\right)\frac{Wr^3}{EI}$$

where:

W is the contact load;

E is the Young's elastic modulus; and

I is the area moment of inertia of ring cross section for the at least one loading planetary roller.

13.   The planetary traction drive transmission of claim 12 wherein the geometry coefficient μ_G is set to closely match the change in a maximum

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available traction coefficient μ_T as the load changes and the geometry coefficient μ_G corresponding to the diametrical reduction of the at least one loading planetary roller is characterized by

$$\mu_G = \tan\left\{\frac{1}{2} Arc\cos\left[\frac{(R_1 + R_2 + e - 2dr)^2 + (R_1 + R_2 - e + 2dr)^2 - 4e^2}{2(R_1 + R_2 + e - 2dr)(R_1 + R_2 - e + 2dr)}\right]\right\}$$

14.   The planetary traction drive transmission of claim 13 wherein the maximum geometry coefficient μ_{Gmax} is set to be no greater than the maximum possible traction coefficient μ_T and the maximum geometry coefficient μ_{Gmax} is characterized by

$$\mu_{G\max} = \tan\left\{\frac{1}{2} Arc\cos\left[1 - 2\left(\frac{e}{R_1 + R_2}\right)^2\right]\right\}$$

15.   The planetary traction drive transmission of claim 14 wherein radii for the at least one supporting planetary roller is characterized by

$$\overline{R}_3 \sin(\alpha_5 - \alpha_3) + \overline{R}_4 \sin(\alpha_5 - \alpha_4) = 0$$

$$\overline{R}_3 \cos(\alpha_5 - \alpha_3) + \overline{R}_4 \cos(\alpha_5 - \alpha_4) + \overline{R}_5 = 0$$

where:

α_5 = the azimuth position of the at least one loading planetary roller;

α_3 = the azimuth position of one of the at least one supporting planetary roller;

α_4 = the azimuth position of the other of the at least one supporting planetary roller;

$\overline{R}_3 = \dfrac{R_1 \cdot r_3}{R_1 + r_3}$ where R_1 is the radius of the second cylindrical

raceway of the sun roller member and r_3 is radius of one of the at least one supporting planetary roller;

$\overline{R}_4 = \dfrac{R_1 \cdot r_4}{R_1 + r_4}$ where R_1 is the radius of the second cylindrical

raceway of the sun roller member and r_4 is the radius of the other at least one supporting planetary roller;

$$\overline{R}_5 = \frac{R_1 \cdot r_5}{R_1 + r_5}$$ where R_1 is the radius of the second cylindrical

raceway of the sun roller member cylindrical raceway and r_5 is the radius of the at least one planetary roller;

16. The planetary traction drive transmission of claim 15 wherein the geometric relationship between the radius and azimuth position of each of the at least one supporting planetary roller is characterized by

$$r_i = \frac{R_2^2 - R_1^2 - e^2 + 2eR_1 \cos\alpha_i}{2(R_2 + R_1 - e\cos\alpha_i)}$$

Where:

α_i = the azimuth position of the at least one planetary roller

r_i = the radius of one of the at least one planetary roller

17. The planetary traction drive transmission of claim 16 wherein the moment of inertia of the material from which the at least one loading planetary roller is made allows the loading planetary roller to deform such that the change in the geometry coefficient either matches or nearly matches the change of maximum available traction coefficient due to changes in the contact load between the at least one loading planetary roller and the outer ring member, and between the at least one loading planetary roller and the sun roller member.

18. The planetary traction drive transmission of claim 17 wherein the first cylindrical raceway and the second cylindrical raceway are slightly crowned.

19. The planetary traction drive transmission of claim 17 wherein the third cylindrical raceway of the at least one loading planetary roller and at least one of the fourth raceways of the at least one supporting planetary roller is slightly crowned.

20. A method of transmitting rotational motion and torque comprising the steps of:

installing a sun roller member into an outer ring member such that the sun roller member is eccentric to the outer ring member and a convergent wedge gap is formed between the sun roller member and the outer ring member;

using one of a material or a combination of materials for at least one

5　　　　　　loading planetary roller which has sufficient flexibility to create adaptive frictional self-loading between the at least one loading planetary roller and the outer ring member and the at least one loading planetary roller and the sun roller member;

installing the at least one loading planetary roller member into the

10　　　　　convergent wedge gap such that the at least one loading planetary roller member is between and in contact with the sun roller member and the outer ring member;

installing the at least one supporting planetary roller to support the sun roller member on a carrier member; and

15　　wedging the at least one loading planetary roller member between the outer ring member and the sun roller member by rotation of either one of the sun roller member or outer ring member such that rotation and torque is transmitted between the outer ring member and the sun roller member.
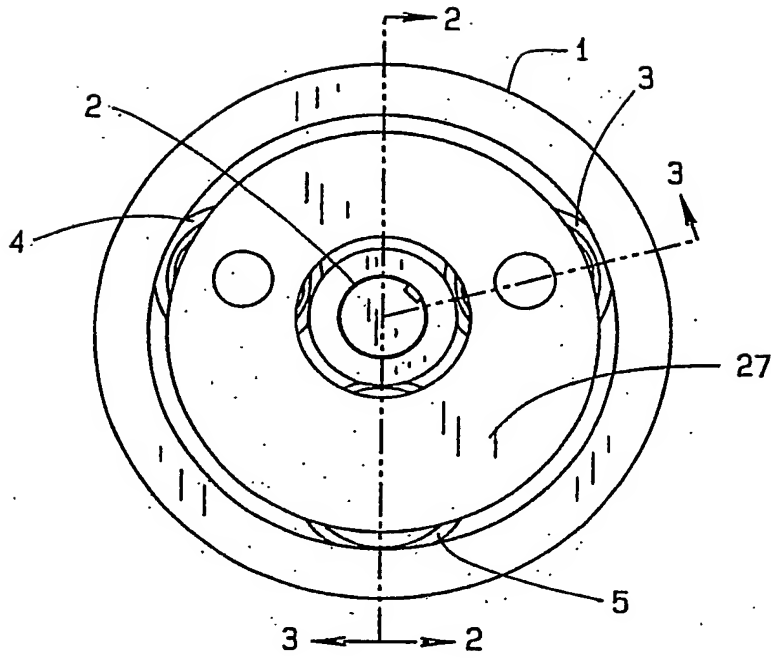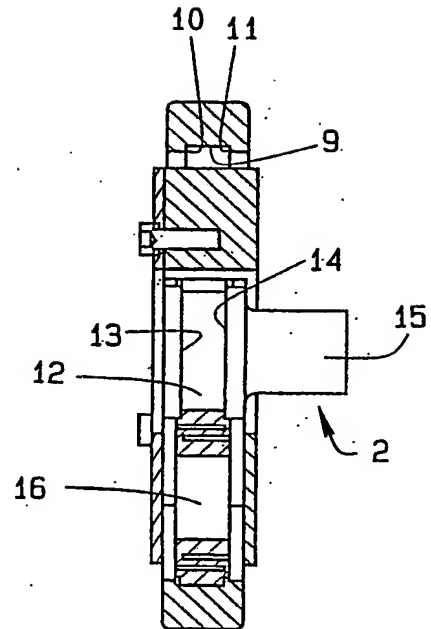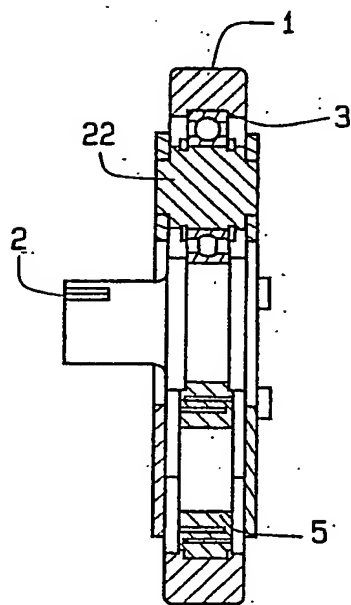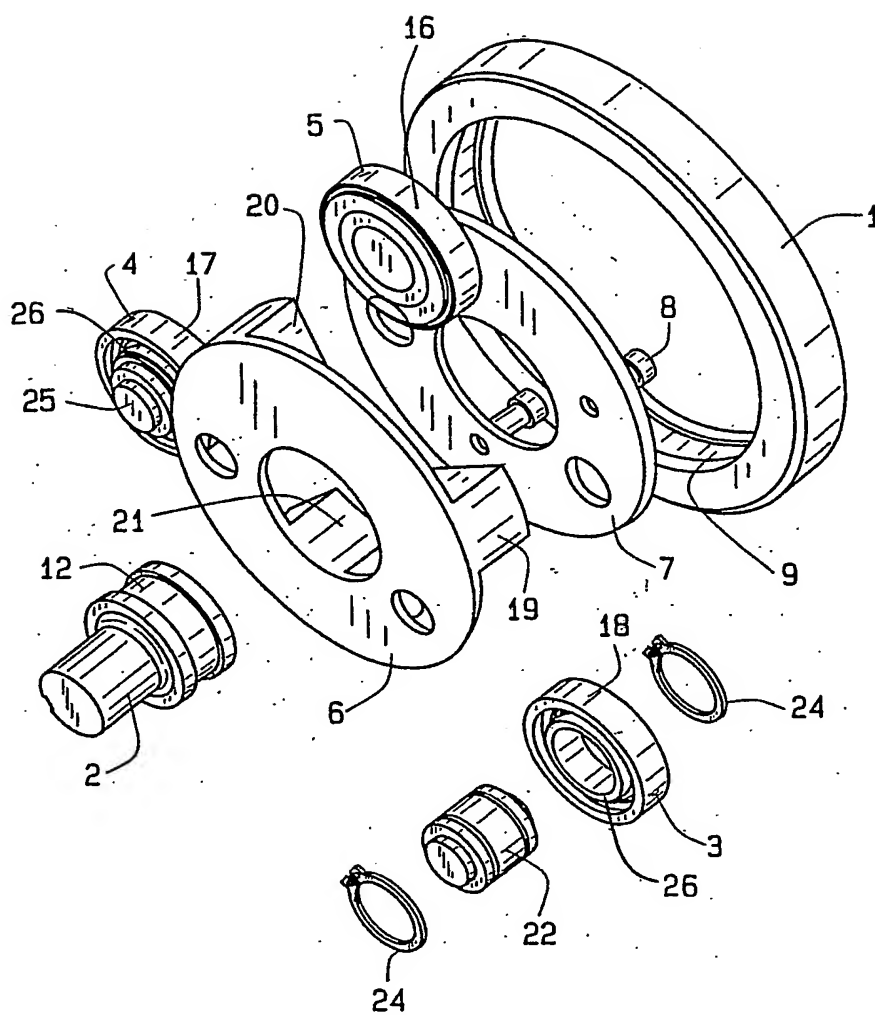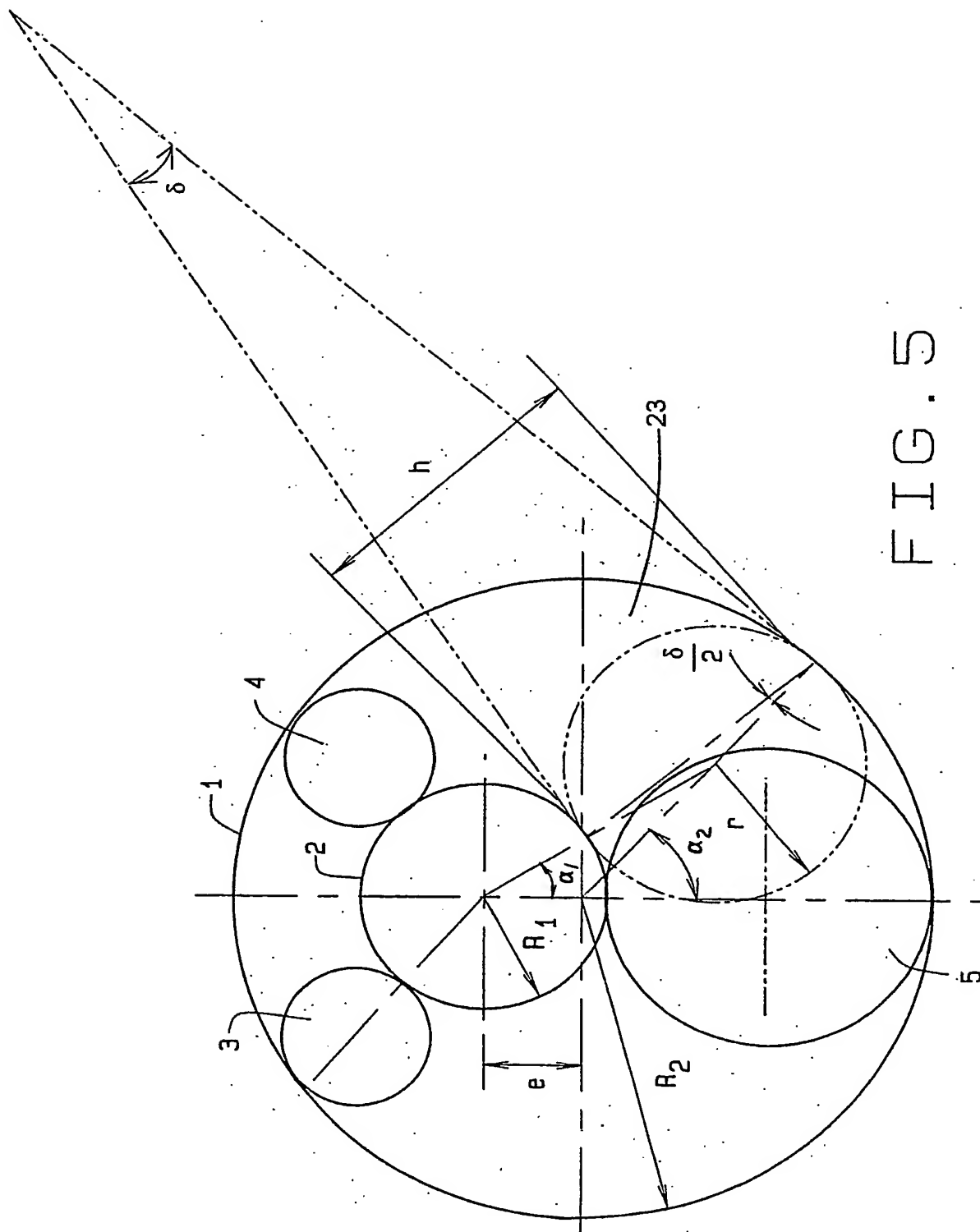
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FIG. 1

FIG. 2

FIG. 3

FIG.4

FIG. 5

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FIG.6

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FIG. 7
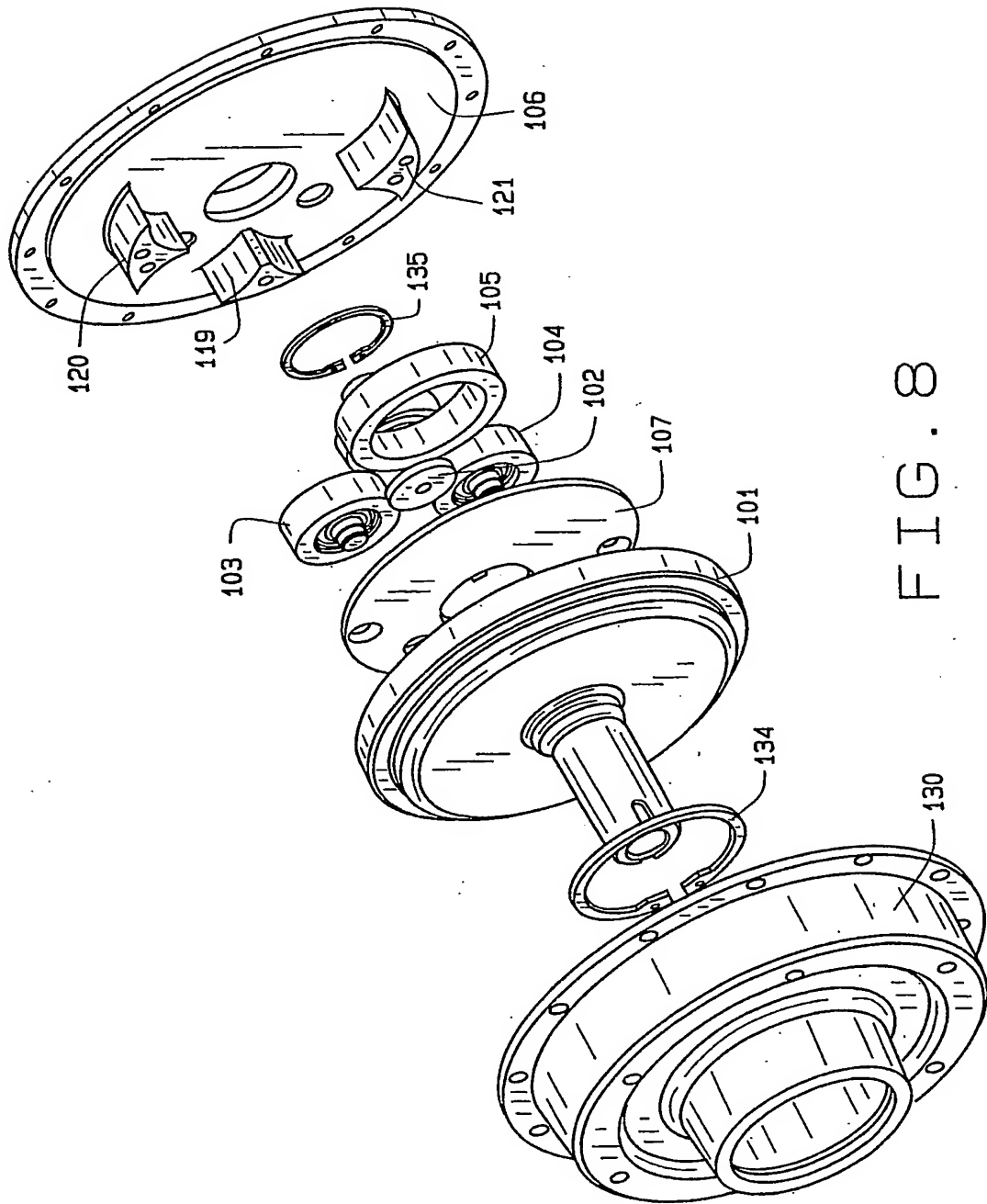
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FIG.8

FIG.9

FIG.10

FIG.11
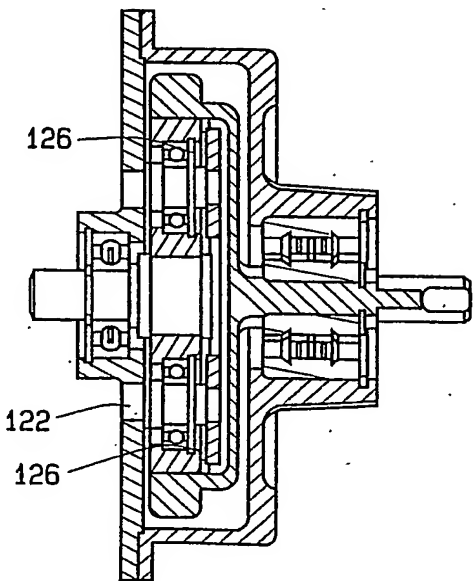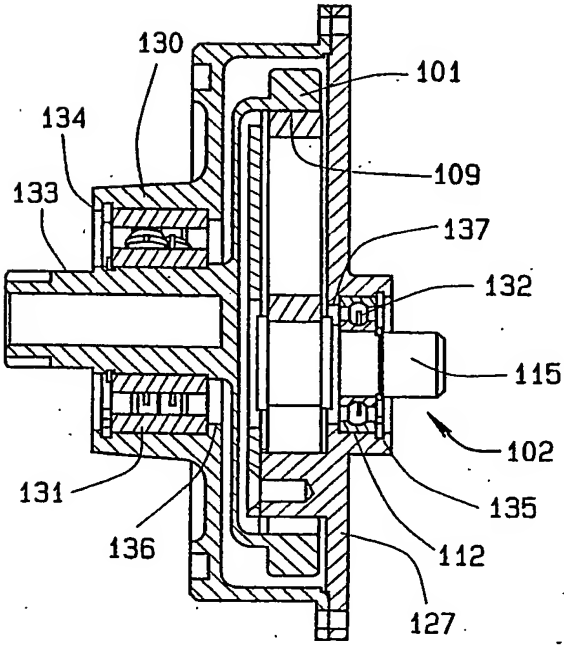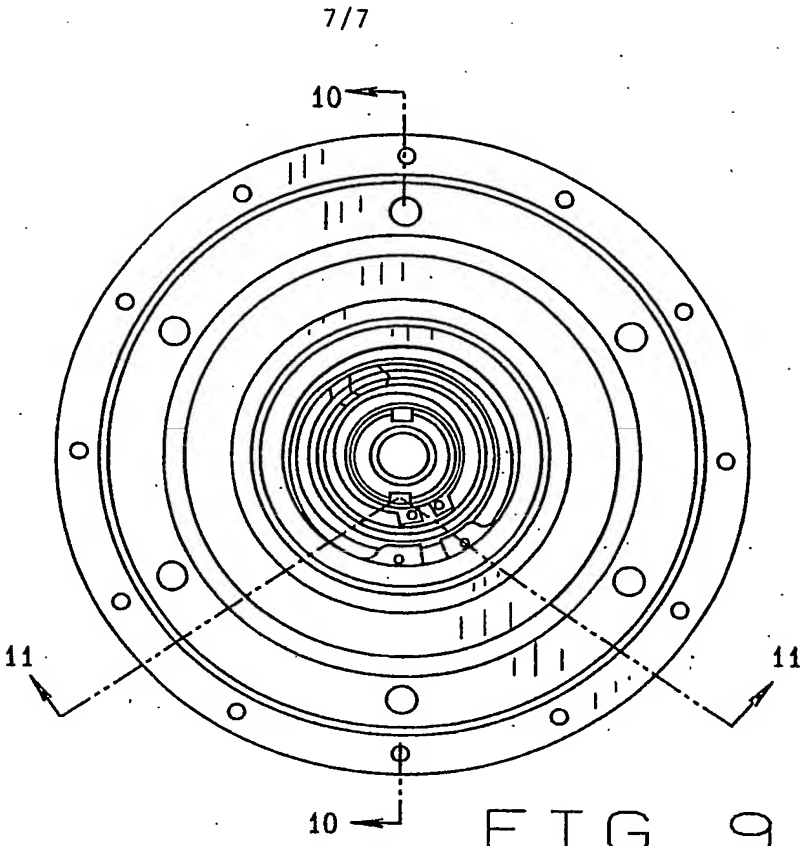
INTERNATIONAL SEARCH REPORT

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7   F16H13/08

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC 7   F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 3 945 270 A (NELSON BERTEL S ET AL) 23 March 1976 (1976-03-23) cited in the application column 3, line 63 –column 4, line 9; figures 1-3 column 5, line 13-45 column 6, line 33-68	1-5,20
A	column 7, line 35-41	6,7
X	PATENT ABSTRACTS OF JAPAN vol. 1999, no. 03, 31 March 1999 (1999-03-31) & JP 10 331933 A (SANYO ELECTRIC CO LTD), 15 December 1998 (1998-12-15) abstract	1-5,20

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[X] Further documents are listed in the continuation of box C.

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Date of the actual completion of the international search	Date of mailing of the international search report
7 May 2003	19/05/2003

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European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Szodfridt, T

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C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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A	EP 0 856 462 A (NSK LTD) 5 August 1998 (1998-08-05) cited in the application page 5, line 50-54; figures 3,4,7,16,17 page 6, line 21 -page 8, line 31 page 13, line 57 -page 14, line 54	1-7
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A	US 4 481 842 A (NELSON BERTEL S) 13 November 1984 (1984-11-13) cited in the application column 2, line 28 -column 4, line 11; figures 1,2	1-5
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